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Subsynchronous Shaft Vibration in an Integrally Geared Expander-Compressor due to Vortex Flow in an Expander

Daisuke Hirata
Acting Manager, Engineering & Design Division
Mitsubishi Heavy Industries Compressor Corporation
Hiroshima, Japan

Naoyuki Nagai
Research Manager, Vibration Research Department
Mitsubishi Heavy Industries, Ltd
Sagamihara, Japan

Hirotaka Higashimori
Senior Chief Engineer
Second Engineering Development
MHI Solution Technologies CO., Ltd
Nagasaki, Japan

Daisuke Hirata is a senior design engineer in Compressor Designing Section in the Engineering and Design Division, Mitsubishi Heavy Industries, Compressor Corporation, in Hiroshima, Japan. He has 9 years experience in designing and developing compressors.
Mr. Hirata has B.S and M.S degrees (in Science) from Hiroshima University.

Naoyuki Nagai is a Research Manager in the R&D Center, at Mitsubishi Heavy Industries, Ltd., in Sagamihara Japan. He has 31 years experience in R&D and troubleshooting for rotor dynamics about compressor, steam turbine. Dr. Nagai received his B.S. /M.S. degrees(in Mechanical Engineering,1983/85) from Kyushu Institute of Technology and Ph.D. degree (in Mechanical Engineering) from Hiroshima University.

Hirotaka Higashimori is a Technical Expert in Engineering Development Section, MHI Solution Technologies Co. Ltd., in Nagasaki Japan. He had 30 year experience in R&D, especially in airodynamics of the Radial Turbomachineries from High Transonic Centrifugal Compressor, Industrial Compressor and Radial Expansion Turbine ,Turbo-Charger for marine use and Automotive Turbocharger to Low Subsonic Centrifugal Fan, in Nagasaki R&D Center Mitsubishi Heavy Industries Ltd.. And he has another 8 years experience of R&D on the same turbomachineries. Dr. Higashimori received his B.S. /M.S. degrees(in Mechanical Engineering,1977/79) from Kyushu University and Doctor degree (in Mechanical Engineering) from Kyusyu University.

ABSTRACT
Subsynchronous shaft vibration was observed in an integrally geared expander-compressor when the machine was operated with a partial load in the course of plant start up. The root cause of the synchronous shaft vibration was identified, by means of CFD analysis, as the vortex flow which was generated in the downstream piping of the gas expander wheel.

OEM installed an object, called “vortex breaker”, in the piping in order to eliminate the excitation force of the vortex flow, and as the result, the subsynchronous shaft vibration disappeared.

This paper provides the detailed shaft vibration data, root cause analysis, countermeasure and the result from the countermeasure.

INTRODUCTION
Vortex flow in a hydraulic turbine is well known as one of the most challenging phenomenon to be controlled. Vortex flow usually happens under a partial load operation due to the relatively higher swirl flow at the outlet of the wheel, and it sometimes becomes a trigger of high noise and/or high vibration. Therefore, many researches on the vortex flow in draft tube have continued in the industry of hydraulic turbine [1, 2, 3]. However, less publication has reported such phenomenon on gas expanders.

This paper introduces the subsynchronous shaft vibration induced by vortex flow in the gas expander of an integrally geared compressor-expander for a nitric acid plant.

MACHINE CONSTRUCTION AND SPECIFICATION
The integrally geared compressor-expander consists of two compressor stages and two expander stages with a gearbox. The compressor 1^{st} stage and expander 2^{nd} stage are equipped in no.1 pinion rotor, and the compressor 2^{nd} stage and expander 1^{st} stage are equipped in no.2 pinion rotor [Fig.1]. This construction, that is one pinion has a compressor impeller and an expander wheel, provides the lower power transmission on gears and lower mechanical losses. Table 1 shows the specification of the machine.

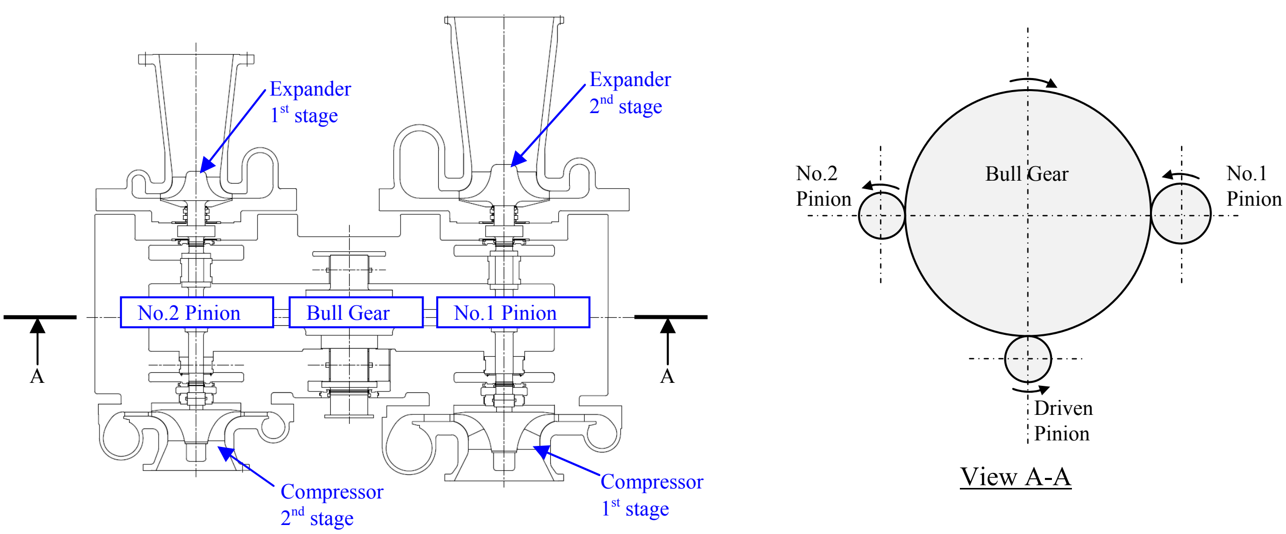


Fig.1 Integrally Geared Compressor-Expander

Table.1 Specification of Integrally Geared Compressor-Expander

Driven Pinion	
Speed	9345 rpm
Bull Gear	
Speed	1000 rpm
Diameter	2244 mm
No.1 Pinion	
Speed	9679 rpm
Compressor Power	3275 kW
Expander Power	3889 kW
No.2 Pinion	
Speed	11292 rpm
Compressor Power	3695 kW
Expander Power	3427 kW

SHAFT VIBRATION

In the course of plant start up, the no.1 pinion rotor faced a steep increase of shaft vibration when the operating parameters reached a particular condition, and immediately after such a condition was passed, the shaft vibration decreased [Fig.2]. The shaft vibration amplitude at the expander side of approx. 70 µm was much higher than that at the compressor side of 30 µm, and the dominant vibration frequency was subsynchronous component of 45~50 Hz (0.3 times operating speed) [Fig.3], which almost matched with one of natural frequencies of no.1 pinion rotor.

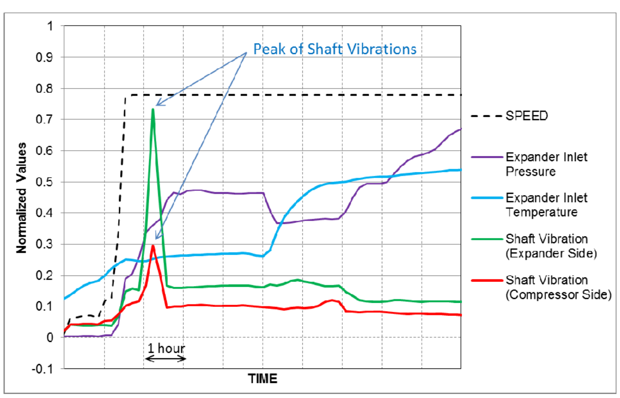


Fig.2 Time Trend Data of Operating Speed, Inlet Pressure/Temperature of Expander 2nd Stage and Shaft Vibration

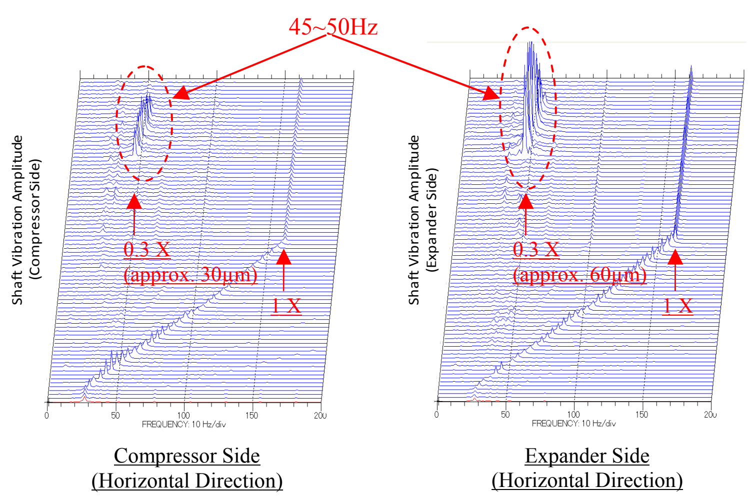


Fig.3 Waterfall Plot during Plant Start Up of No.1 Pinion Rotor

ROOT CAUSE INVESTIGATION

Subsynchronous shaft vibrations in an integrally geared type machine are sometimes observed during machine start up in relation with the transition of loading conditions. For example, when a weight (downward) and a gear mesh force (upward) on a pinion rotor are balanced, a load on journal bearings becomes too small to form a stable oil film by the wedge effect. As the result, subsynchronous shaft vibrations can happen. Therefore the first step for a root cause investigation was to review a rotor-bearing system including the detailed rotor stability analyses [4]. However, any analyses regarding the rotor-bearing system could not suggest a root cause of the problem.

Finally the flow induced excitation force from downstream piping was pointed out as one of the possible cause with reference to past experience of a hydraulic turbine [3]. In order to simulate the flow pattern in the piping downstream of the 2nd stage expander wheel, time transient CFD analysis was conducted. Boundary conditions for the CFD analysis were defined based on the recorded operating parameters, such as pressure/temperature of inlet/outlet and so on, at the timing when the steep increment in shaft vibrations was observed [Fig.4].

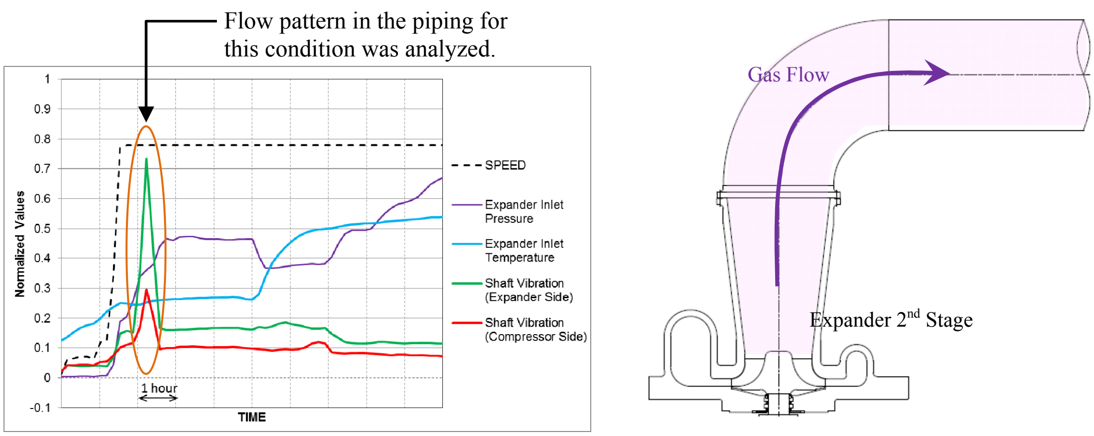


Fig.4 Condition and Objects for CFD Analysis for 2nd Stage Expander Wheel

Fig.5 shows velocity contour and equipollent plane of static pressure in the piping downstream of 2nd stage expander wheel, these figures provide a shape of vortex. Fig.6 shows a transition of pressure contour with time at reference plane shown in Fig.5 viewed from downstream of the expander wheel. The pressure contour clearly shows a core of the vortex which is revolving with time.

Fig.7 shows the pressure distributions in 2nd stage expander wheel. The contours for each section (right side figures in Fig.7) show the circumferential uneven pressure distributions in the 2nd stage expander wheel which can be generated by the vortex in the piping. This uneven pressure distribution causes excitation force on the expander wheel in a direction from higher pressure side to lower pressure side. For example in section A-A, B-B and C-C in the Fig.7, the pressure on left side is relatively higher than that on right side, consequently the excitation force, which can be estimated as the integration of the pressure over the surface area of the wheel, from left side to right side on the expander wheel can be induced.

Additionally, the pressure pulsation at a point in the reference plane was examined by means of frequency analysis to the CFD result, and the spectrum showed a remarkable peak with 52 Hz as shown in Fig.8.
This result suggests that 2nd stage expander wheel must be forced by the excitation force with frequency of 52 Hz. This frequency agrees with the dominant frequency (45~50 Hz) of the measured shaft vibration of no.1 pinion rotor.

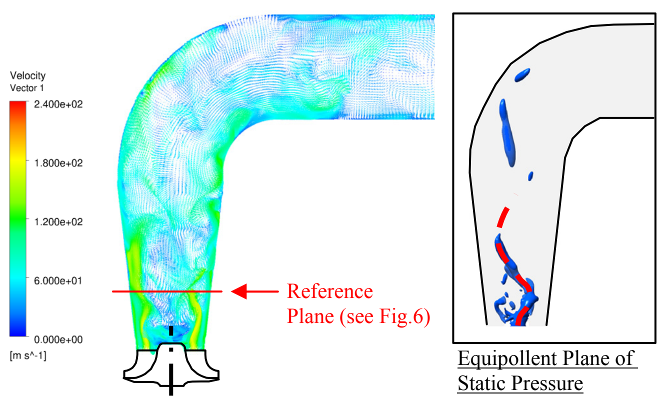


Fig.5 Velocity Contour and Equipollent Plane of Static Pressure in Piping

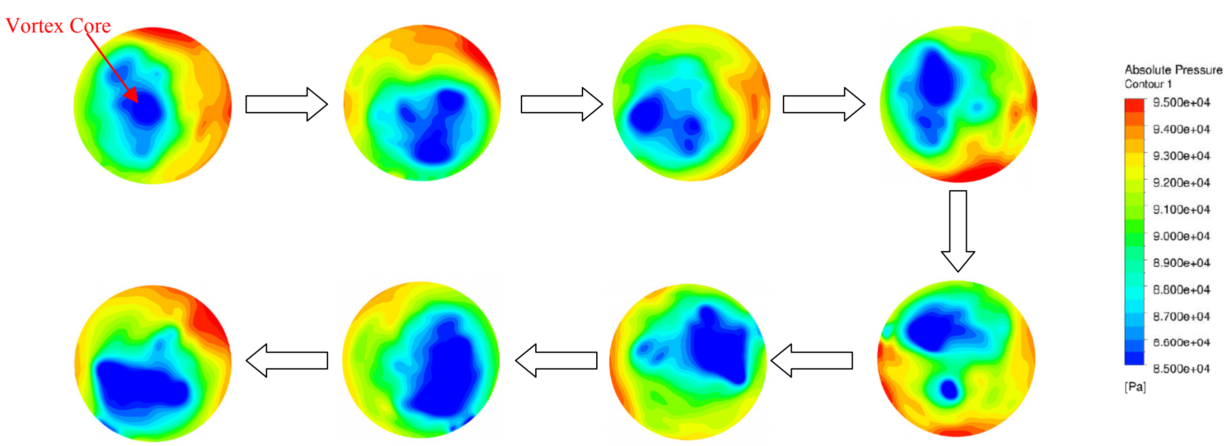


Fig.6 Time Transition of Pressure Contour at Reference Plane in the Piping of Fig.5 viewed from downstream of 2nd Stage Expander Wheel

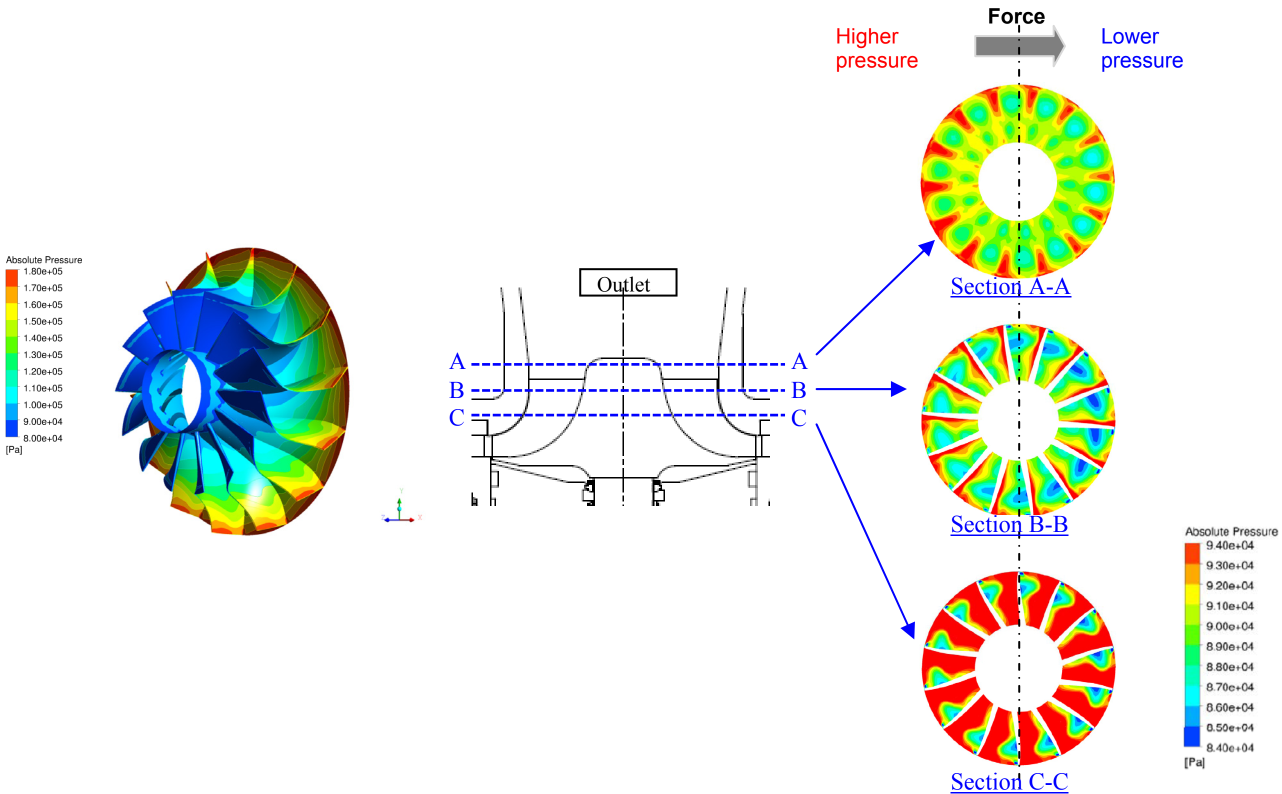


Fig.7 Pressure Distribution in 2nd Expander Wheel

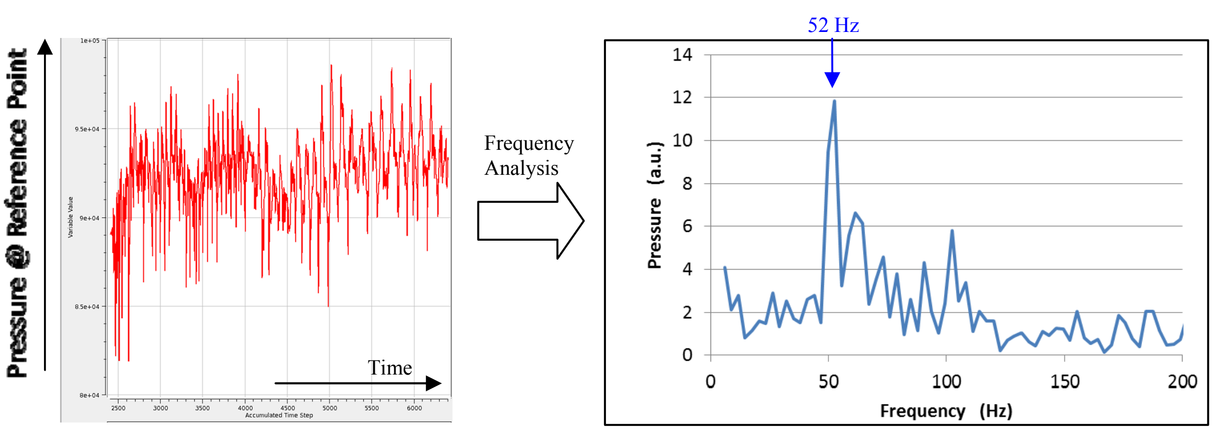


Fig.8 Pressure Pulsation at Reference Point in Downstream Piping from CFD

Furthermore, rotordynamics for no.1 pinion rotor was studied with a consideration of the excitation force on 2nd stage expander wheel. That is ; the non-synchronous rotor vibration responses at rated speed were calculated by applying a unit external force with changing its frequency onto 2nd stage expander wheel in the rotordynamic model [Fig.9]. Fig.10 shows rotor vibration responses verses frequency of the unit excitation force, and it is confirmed that there are some peaks which are corresponding to the natural frequencies or the rotor at around 38 Hz and 50 Hz. As the predicted frequency of the excitation force from a vortex is 52 Hz, the second peak must be amplified by an interaction with the excitation force.

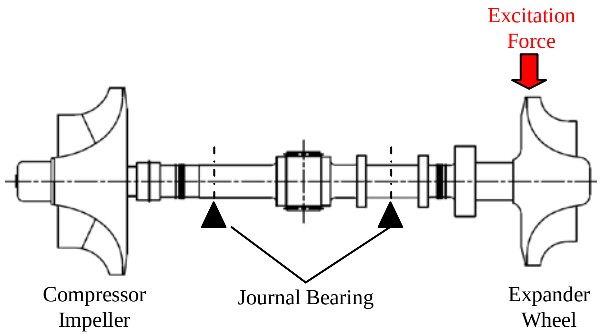


Fig.9 Rotordynamics Model for no.1 Pinion Rotor

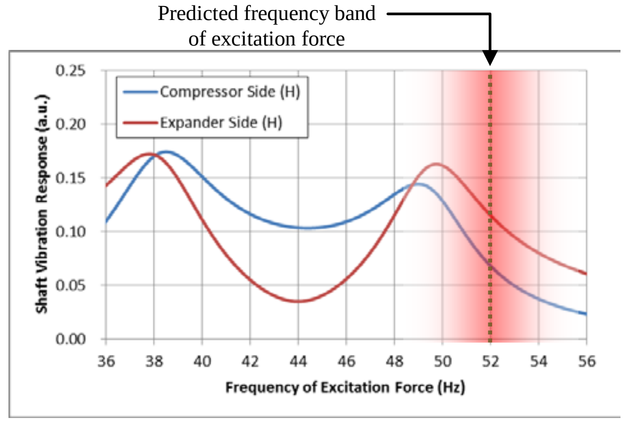


Fig.10 Shaft Vibration Response against Excitation Force on Expander Wheel of no.1 Pinion Rotor

COUNTERMEASURE

In order to eliminate the pressure pulsation due to the vortex, the application of a "vortex breaker [Fig.11]" was evaluated by another time transient CFD analysis. It was expected by the result from CFD that the vortex could be broken and the uneven pressure distribution in the piping could be suppressed as shown in Fig.12.

Fig.13 shows the comparison of pressure distribution in 2nd stage expander wheel between with and without the vortex breaker. It can be confirmed that the uneven pressure distribution [Fig.13 b] is much improved by applying the vortex breaker [Fig.13 a]. Moreover, the frequency analysis from CFD result shows no peak in the pressure spectrum [Fig.14].

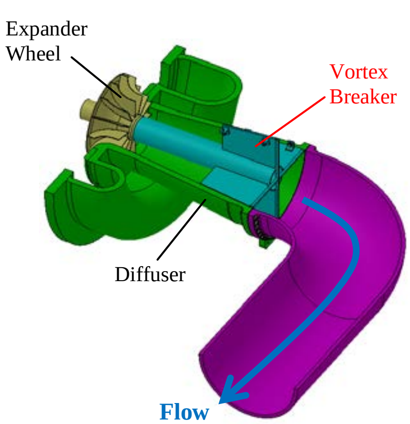
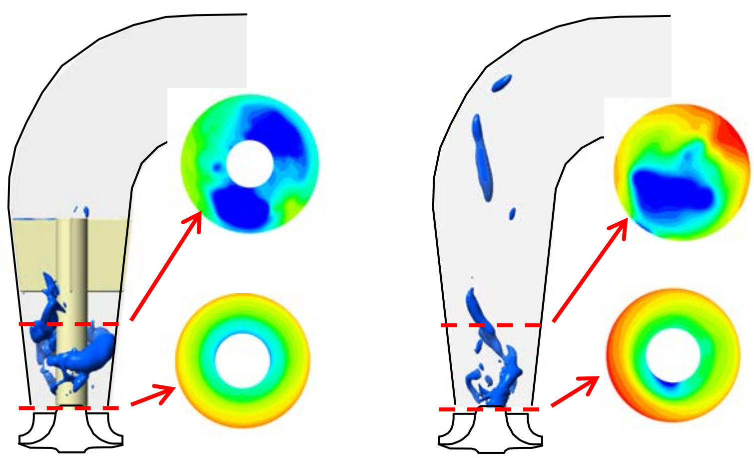


Fig.11 Vortex Breaker



a.) with Vortex Breaker

b.) without Vortex Breaker

Fig.12 Comparison of the Flow and Pressure in the Piping with/without Vortex Breaker

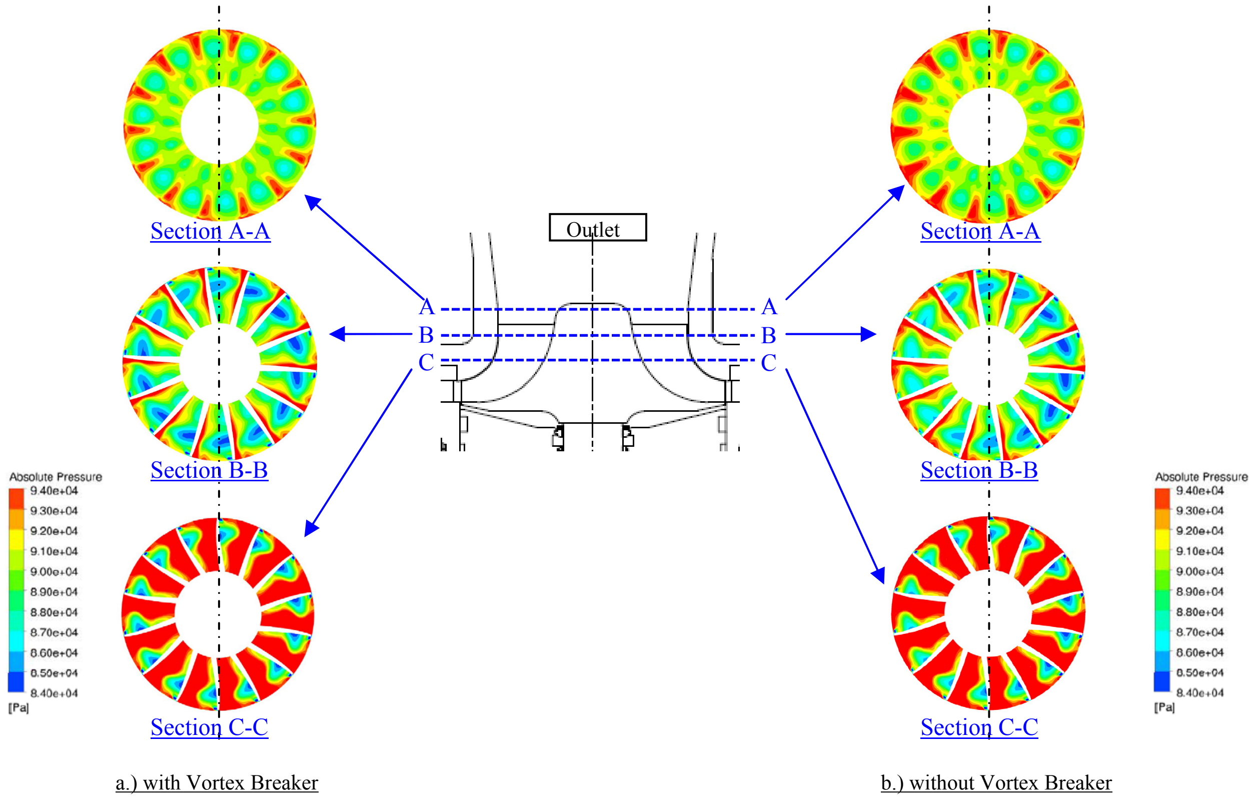


Fig.13 Pressure Distribution on Expander Wheel with/without Vortex Breaker

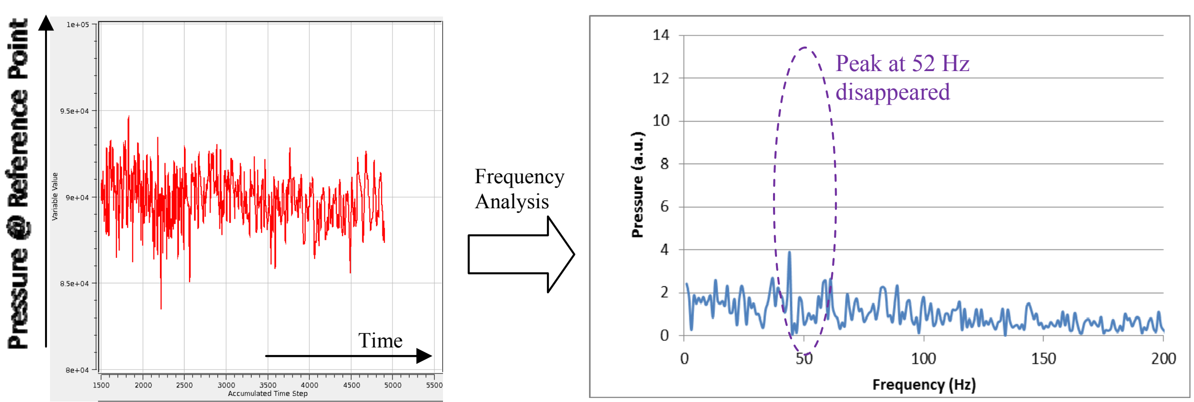


Fig.14 Pressure Pulsation at Reference Point in Downstream Piping with Vortex Breaker from CFD

With reference to the result of the root cause analysis, the OEM manufactured a vortex breaker, and installed it into the actual machine at site in collaboration with both an end-user and a plant engineering company.

Fig.15 shows the time trend data of operating parameters of a verification run after countermeasure was implemented. In this run, the process conditions (pressure and temperature of process gas) were repeatedly (three times) controlled to get the particular condition, in which the high shaft vibration had been observed since initial start up, in order to prove the effectiveness of the vortex breaker. During this operation, the shaft vibration in no.1 pinion on both compressor side and expander side did not show the steep increase and the subsynchronous vibrations at 45~50 Hz also disappeared as predicted by the analyses [Fig. 16]

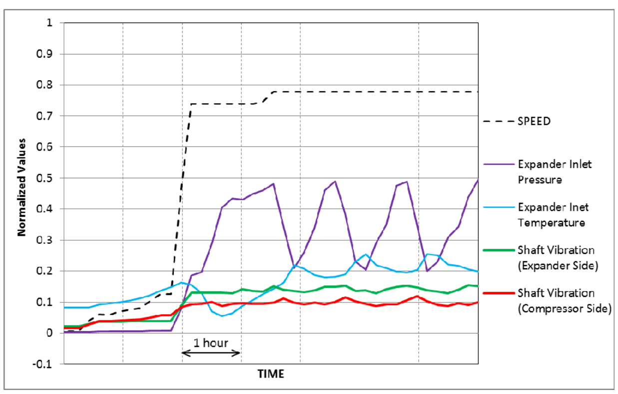
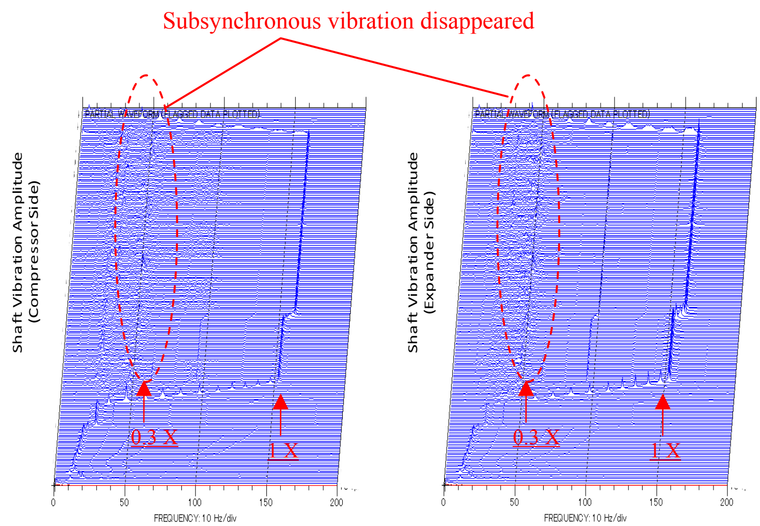


Fig.15 Time Trend Data of Operating Speed, Inlet Pressure/Temperature of Expander 2nd Stage and Shaft Vibration after Countermeasure



Compressor Side (Horizontal Direction)

Expander Side (Horizontal Direction)

Fig.16 Waterfall Plot of No.1 Pinion Rotor after Countermeasure

CONCLUSIONS

- High subsynchronous shaft vibration with in an integrally geared compressor-expander under partial load operation was reported.
- Any analyses regarding rotor-bearing system could not explain the phenomena, and finally flow induced excitation force from downstream piping of expander wheel was reviewed.
- Time transient CFD analysis predicted the presence of vortex flow downstream of expander wheel under partial load operation.
- Pressure pulsation induced by the vortex flow generated an excitation force onto the expander wheel, and its predicted frequency was 52 Hz which well agreed with the dominant frequency (45~50Hz) of measured shaft vibrations.
- Rotordynamic analysis suggested that the shaft vibration at a natural frequency must be amplified by resonance between rotor system and fluid system.
- As the countermeasure, installation of a vortex breaker was proposed and its effectiveness was evaluated by CFD analysis.
- OEM manufactured the vortex breaker and installed it into the machine in site, and subsynchronous shaft vibration under partial load operation disappeared.
- The root cause of high shaft vibration was proven as a vortex flow in the downstream piping of 2nd stage expander wheel by both theoretical analysis (CFD) and measurements on the actual machine in site.
- The challenge for the future project is to clarify the particular condition which can generate the vortex flow in the expander.
- The recommendation so far is to minimize the swirl flow even under the off designed partial load operating condition.

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